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15 METER HOOP-COLUMN ANTENNA DYNAMICS: TEST AND ANALYSIS

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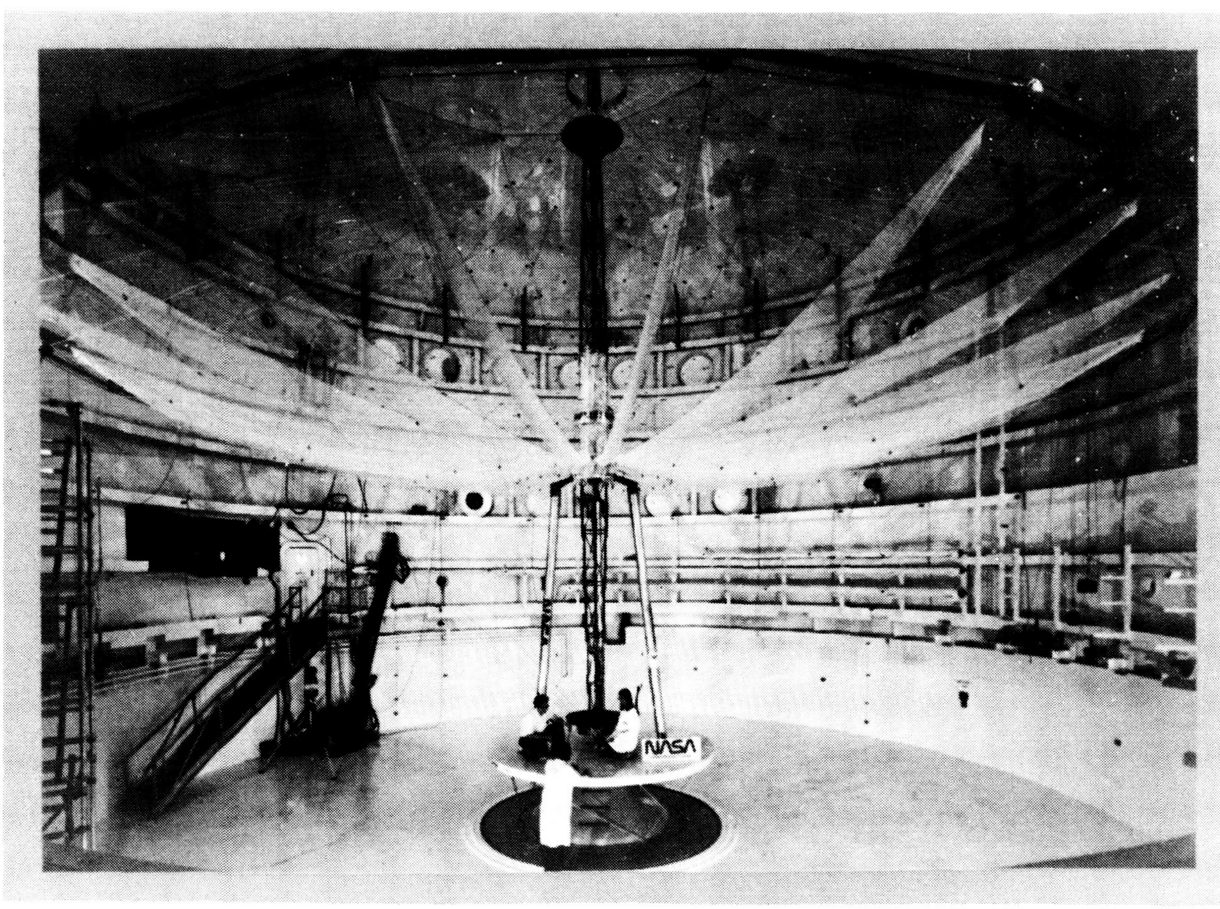
INTRODUCTION

Large Space Antennas (LSA) proposed for communication and remote sensing missions will require accurate dynamic analyses to predict structural vibration behavior and to assess the need for active vibration control. Of the numerous LSA concepts that have been proposed (see for example Refs. 1, 2 and 3), one concept, referred to as the Hoop-Column concept, has been fabricated (Ref. 2). A 15 meter diameter model has been constructed for deployment, electromagnetic and structural testing. The model, shown below, was designed and fabricated jointly by government and industry (Ref. 4). As part of the test program, static and dynamic tests have been performed in the Langley 16 Meter Vacuum Chamber. Results from these tests and comparisons with predicted structural behavior will be described.

15 METER DIAMETER HOOP-COLUMN ANTENNA



OUTLINE

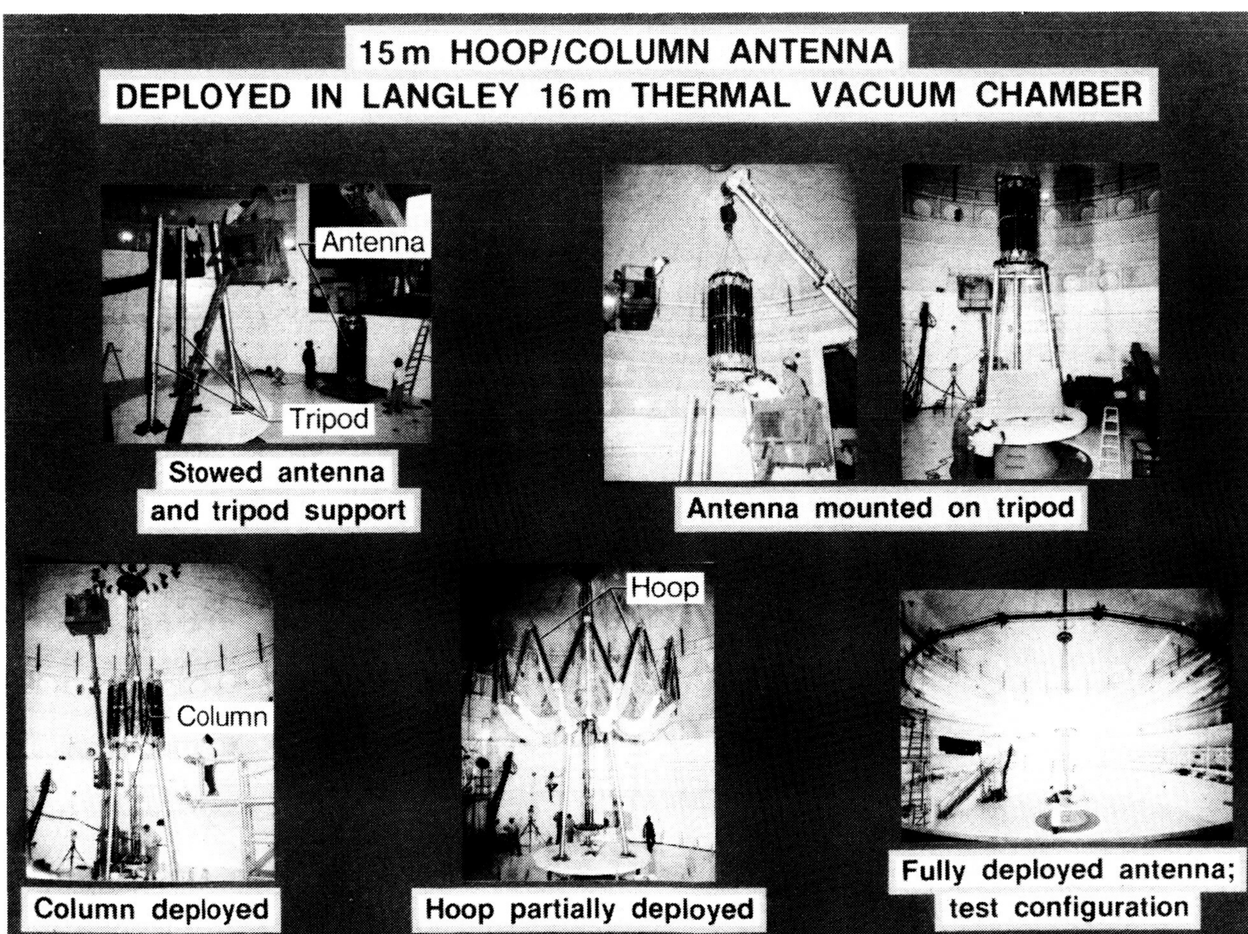
A number of structural tests and analyses have been performed on the 15 meter antenna model. This paper will concentrate on vibration test and analysis results. As indicated in the outline below, a description of the experimental and analytical models will be presented first. Second, test and analysis results will be presented for the antenna mounted on a tripod support. Vibration results have been divided into three areas: global modes which are dominated by overall column bending and hoop motion; hoop modes which are dominated by inplane and out-of-plane hoop (ring) bending; and mesh modes which are localized surface mesh distortions. Results from a simplified analytical model will be presented followed by a description of the antenna vibrations when supported by a pendulum cable.

Description of:

- 15 meter hoop column antenna
- Antenna vibration tests
- Antenna analysis models
- Vibration modes with tripod support
  global, hoop and mesh
- Reduced analytical model
- Global vibration modes
  Cable suspension
- Summary

ANTENNA DEPLOYMENT

The figure below shows the antenna deployment sequence. First, the antenna is mounted on a tripod support in a stowed configuration. Next, the column is deployed by a motor driven cable system enclosed in the column longerons. The column telescopes outward simultaneously from top and bottom. After the column is deployed, the hoop is deployed by eight motors which are mounted in eight of the twenty-four hoop joints. Synchronizing rods are used to maintain uniform deployment of each hoop section. The final deployment step is the actuation of a preload segment at the bottom of the column. This segment extends outward to pretension all of the cables and mesh, thus providing a stable structural configuration. The antenna deploys from a volume of approximately 1 m by 3 m to a deployed size of 15 m in diameter by 9.5 m in height. The feed and feed mast were manually attached after the antenna was deployed.



Stowed antenna and tripod support

Antenna mounted on tripod

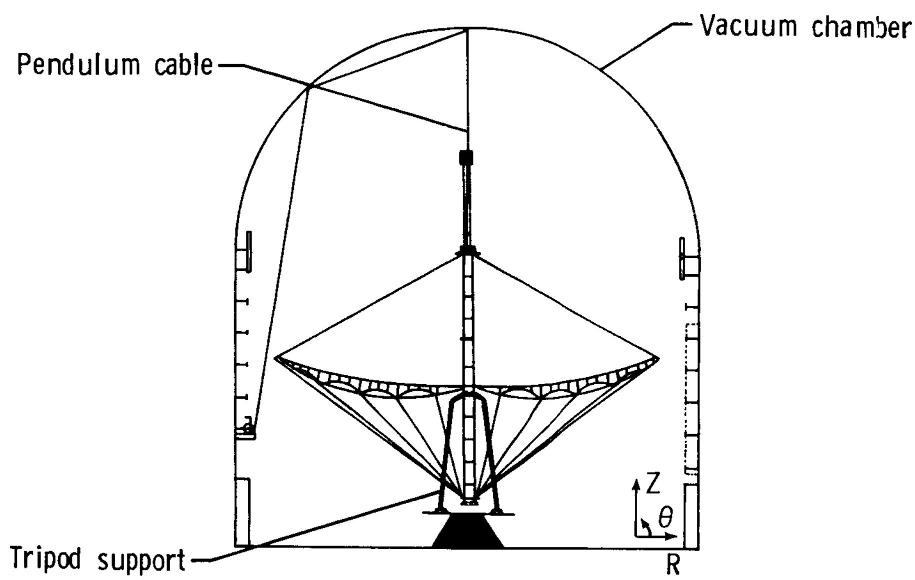
Column deployed

Hoop partially deployed

Fully deployed antenna; test configuration

ANTENNA EXPERIMENTAL SETUP

The antenna was deployed inside a vacuum chamber as shown in the schematic diagram below. The deployment was stopped on several occasions to enable accelerometers and strain gages to be mounted on the column and hoop. A total of 58 servo accelerometers were used to measure the acceleration response of the antenna due to small random and sinusoidal disturbances. In addition, displacement measuring proximity probes were used to measure static deflections and surface mesh vibrations. The total mass of the antenna is 400.1 kg (880.5 lbs) with the center of gravity located at 1.85 m above the hoop. The rotational inertias of the antenna are $I_x = I_y = 11500$ kg-m^2 (101800 lb-s^2-in) and $I_z = 8160$ kg- m^2 (72200 lb-s^2-in).



Antenna properties

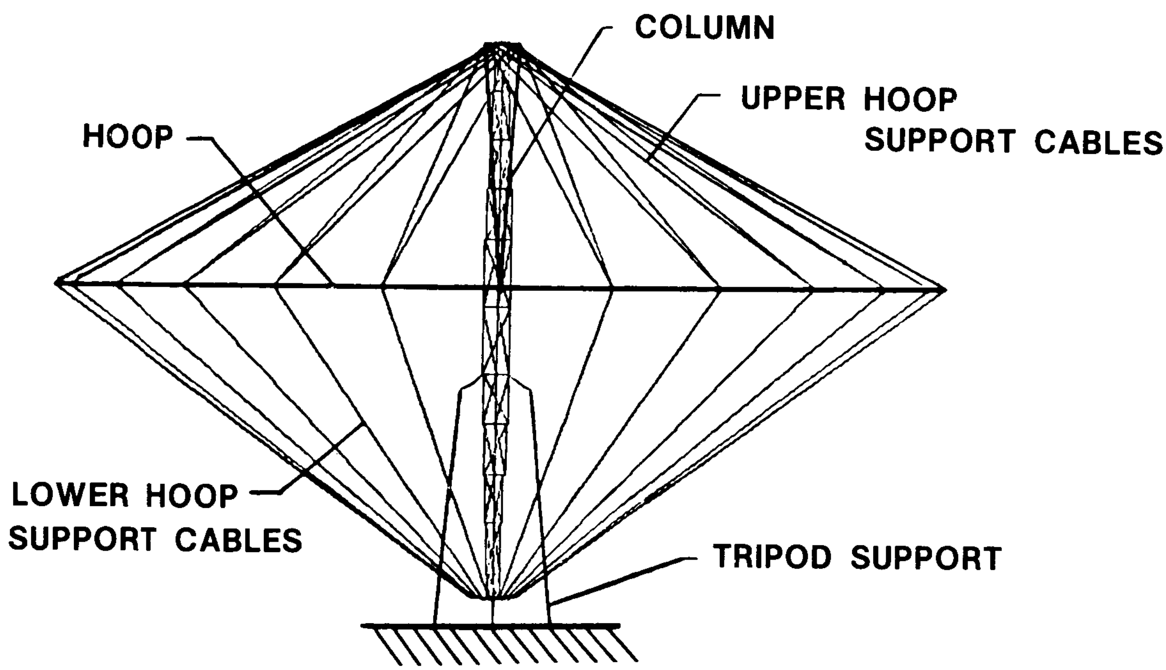
Overall dimensions:

Diameter = 15 M (590.5 in.)
Height = 12.8 M (504 in.)

Elements	Mass (weight)
Hoop (graphite)	131.8 kg (290.1 lbs)
Column (graphite)	135.8 kg (298.7 lbs)
Surface mesh (gold plated molybdenum wire)	10.7 kg (23.6 lbs)
Cables (graphite and quartz)	2.1 kg (4.7 lbs)
Feed mast (steel)	11.5 kg (25.4 lbs)
Simulated feed weight	108.2 kg (238 lbs)
Total	400.1 kg (880.5 lbs)

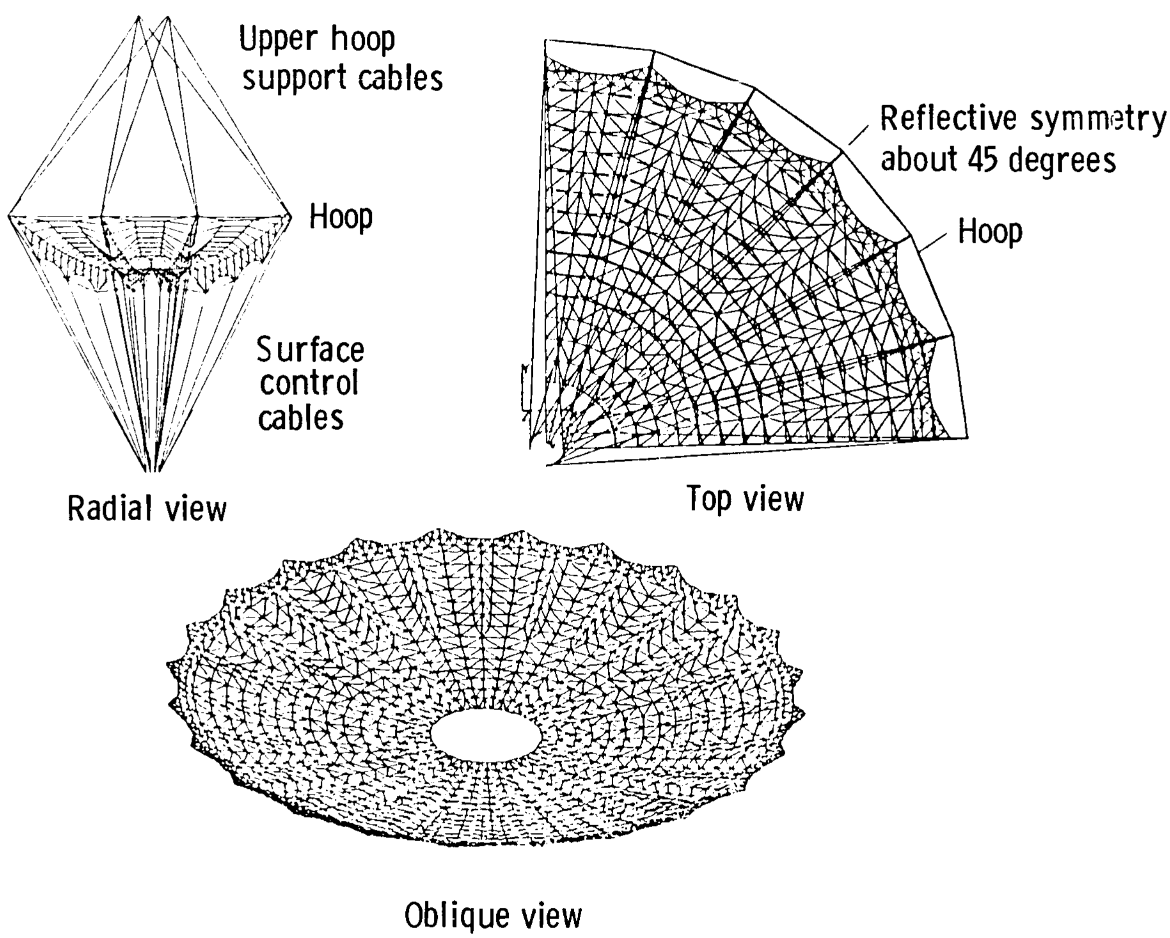
ANTENNA FINITE ELEMENT MODEL: WITHOUT SURFACE

The antenna has been modeled with the Engineering Analysis Language finite element program described in Ref. 5. The hoop, column and hoop support cables were modeled first. The model shown below includes the flexibility of the tripod support. Preliminary vibration tests were performed prior to surface mesh installation to verify the antenna analytical model without the surface. These preliminary tests showed the need for modeling the tripod and including the rotational inertia of the column. The differential stiffness due to tension/compression loads in the members was included an all analyses. In addition, gravity loading was modeled to correlate with ground vibration test results.



ANTENNA FINITE ELEMENT SURFACE MODEL

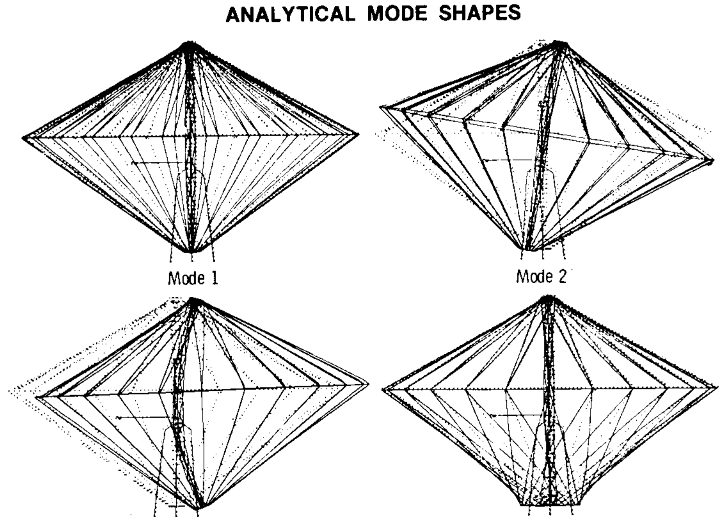
The antenna surface consists of four paraboloidal apertures shaped by pretensioned cables and mesh. Since twenty-four gores are present, the smallest representative element of the antenna surface is a three gore model as shown in the upper left of the figure below. Reflective symmetry permits one full aperture to be modeled as shown in the upper right six gore model. Repeated application of rotational symmetry yields the full surface model shown in the bottom figure. (Only mesh elements are shown for clarity.) The surface model was merged with the hoop and column model shown previously to permit analysis of the full antenna. The analytical model without the feed and feed mast contained 286 beam elements, 4664 rod elements and 2880 triangular mesh elements. A total of 2096 grid points and 8816 degrees of freedom were used in the analysis.



FREQUENCIES OF FIRST FOUR VIBRATION MODES

Test results showed the fundamental frequency of the antenna was 0.077 Hz when supported on the tripod. This mode was dominated by torsion of the hoop. As indicated in the table below, the second mode shape occurred at 0.704 Hz and consisted of hoop rocking and column bending. The number of modes at this frequency is indicated in parenthesis to be two since this mode shape can occur in two orthogonal planes. Although fabrication asymmetries usually yield two distinct frequencies, test results indicated no measurable difference in frequency when excited in different planes of vibration. The next mode shape was dominated by hoop inplane translation and column bending. Again two modes occur at the frequency of 1.76 Hz. The last mode indicated below is characterized by torsional motion of the lower column at 3.06 Hz. These four modes were found to be the dominant global modes when supported on the tripod. Initial analysis results showed significant errors in frequency when compared to the test data. The properties in the initial analysis were based on fabrication drawings, which can often lead to overestimates of member stiffnesses by neglecting the joints used in the assembly process. Thus, static tests of the assembled antenna were used to measure the effective stiffness of the hoop and column for refinement of the analytical model.

ANALYTICAL MODE SHAPES

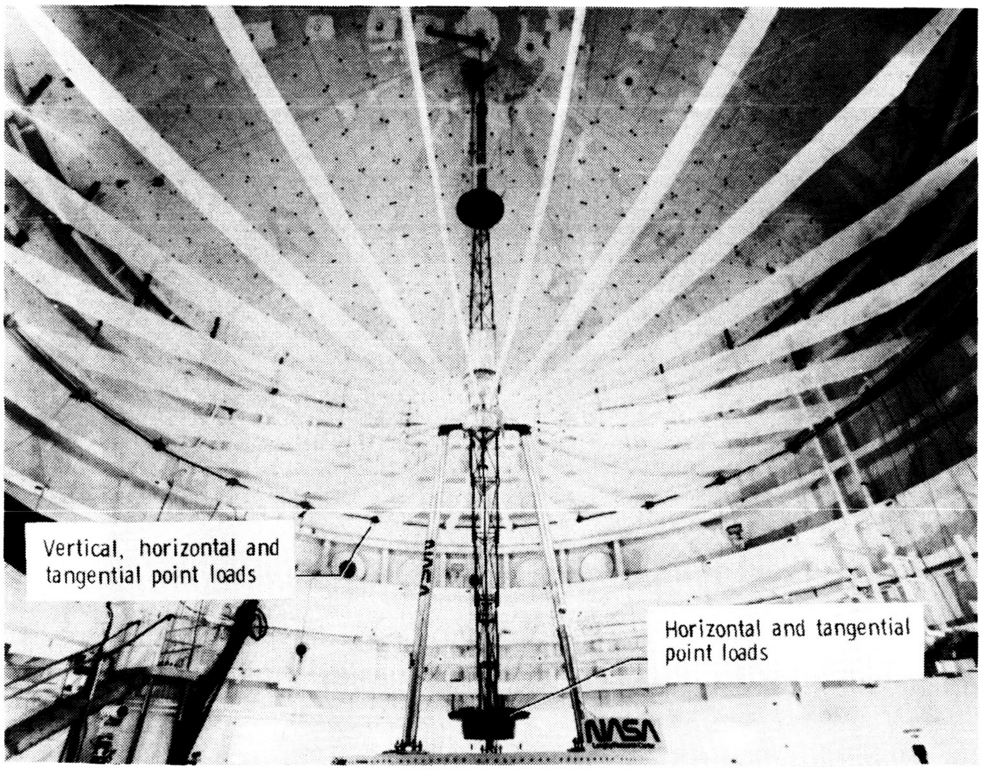


ORIGINAL ANALYSIS FREQUENCIES

Mode	Original analysis	Test	Mode shape
	Frequency (HZ)		
1	0.092	0.077	Hoop torsion
2 (2)	1.60	0.704	Hoop rocking/column bending
3 (2)	2.80	1.76	Hoop inplane/column bending
4	3.20	3.06	Lower column trosion

STATIC TESTS FOR ANALYTICAL MODEL REFINEMENT

Since component and subassembly tests of joints and other antenna parts could not be performed, the entire antenna was loaded to measure the effective member stiffnesses. The figure below indicates locations where point loads were applied to the antenna. Vertical, torsion and radial loads were applied to the hoop whereas only radial and torsional loads were applied to the base of the column. The deflection of the antenna was measured at the point of loading and computed using the analytical model. Comparisons between test and analysis indicated the analysis overestimated the stiffness by 17 percent for column bending and 10 percent for hoop torsion. Mass measurements of the antenna were also performed and used to update the analytical model. The analytical model was modified by adjusting the stiffness of column longeron and diagonal members and by adjusting the stiffness of the column to tripod attachment. The analytical model, refined using the static test data, was then used to recompute the first four vibration modes of the antenna.



REFINED ANALYTICAL MODEL RESULTS

The table below lists the analysis and test frequencies after the antenna model was refined. Good frequency agreement was obtained. This indicates static test data should be used to refine the analytical models of large space structures particularly since the accuracy of static data increases as the flexibility of the structure increases. Also listed in the table are the measured frequencies and damping of the first four modes of the antenna. The damping is somewhat higher than usually found in spacecraft perhaps due to the deployable joints and the tripod interface. The damping increases when tested in ambient air. Unlike panel structures, the antenna shows very small decreases in frequency when tested in ambient air. This indicates large lattice structures can be tested in ambient air without significant changes in vibration frequencies of the global modes.

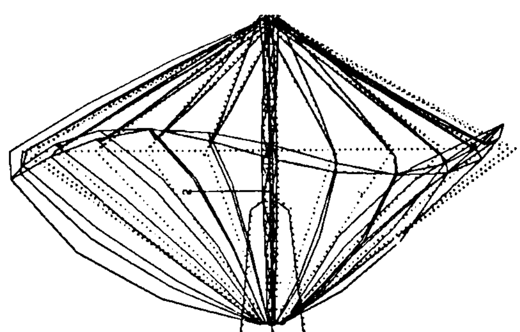
Mode	Refined analysis	Test			
		Vacuum		Ambient air	
	F (HZ)	F (HZ)	C/CR (%)	F (HZ)	C/CR (%)
1	0.077	0.077	1.9	0.076	3.8
2 (2)	0.697	0.704	3.8	0.700	4.3
3(2)	1.73	1.76	3.2	1.75	3.3
4	3.18	3.06	0.84	3.10	1.3

HOOP VIBRATION MODES

Although the antenna vibration is dominated by the global modes shown previously, a number of more localized modes exist. Out-of-plane ring bending of the hoop produces 12 modes from 6.3 to 10.1 Hz in the analysis. In addition, another 14 inplane ring bending modes occur from 10.7 to 14.4 Hz. As shown in the figure below, the modes shapes of the hoop modes involve rotation of the hoop joints. Since these hoop joints have some unknown effective rotational stiffness, mode for mode comparisons between test and analysis is almost impossible. Nevertheless, modeling the joints as being pinned in the out-of-plane direction and being rigid in the inplane direction, has yielded a frequency spectrum which agrees quite well with the test data. Some nonlinearity in the hoop modes has been found when testing at different force levels. It is felt that a mechanical locking mechanism to secure the joint in the deployed configuration or a method to completely pin the hoop joints would be useful in reducing nonlinearities and thus, permit better simulation of the hoop response.

Out-of-plane bending

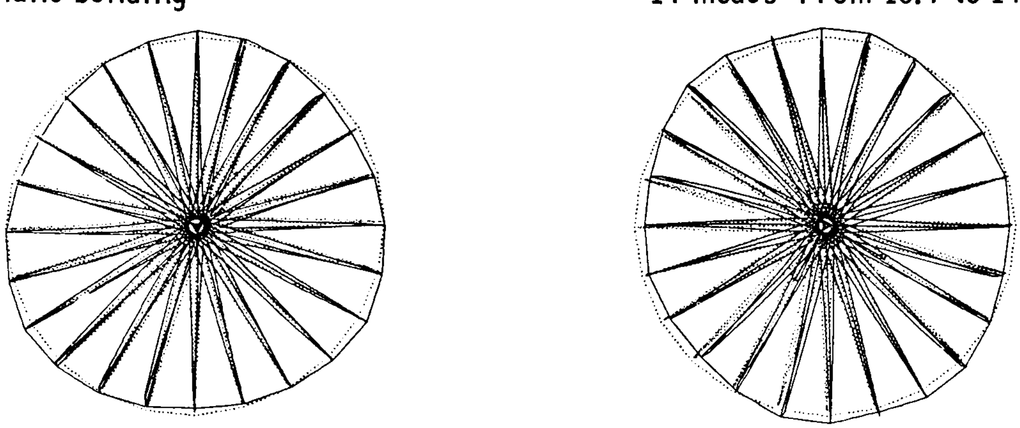
12 modes From 6.3 to 10.1 HZ



Typical out-of-plane hoop modes

Inplane bending

14 modes From 10.7 to 14.4 HZ

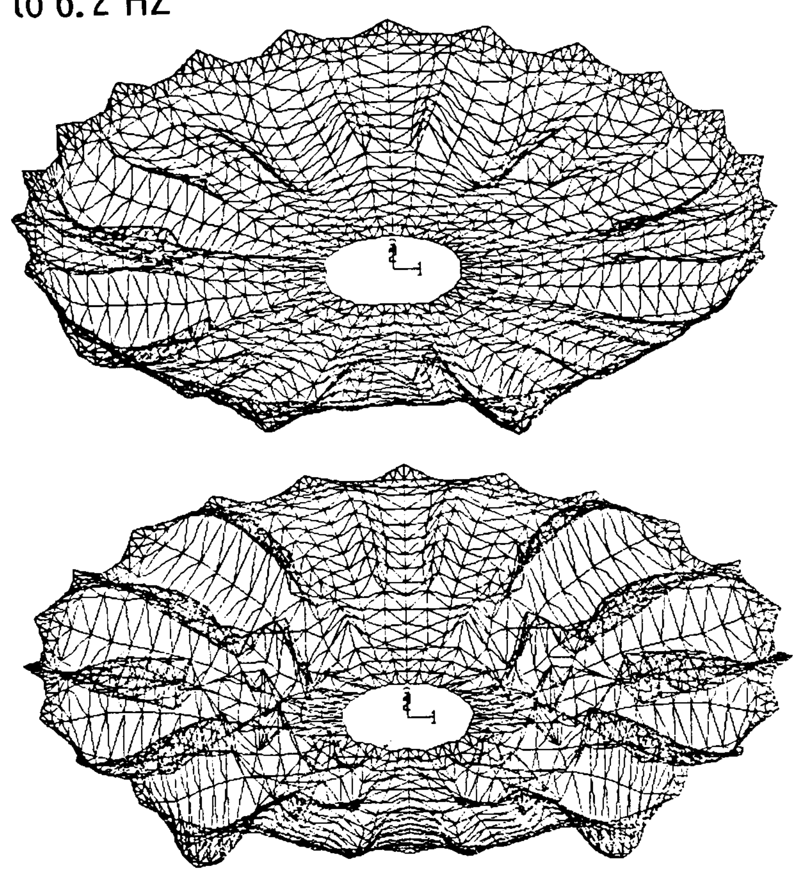


Typical in-plane hoop modes

MESH VIBRATION MODES

The analytical model predicts 70 vibration modes dominated by surface mesh displacements from 4.1 to 6.2 Hz. Shown below are typical mode shapes predicted by the analysis. Experimentally, these modes have been found to be highly damped and coupled. Although a frequency spectrum in this range can be found by testing the antenna, the data has not been successfully reduced into a set of recognizable mode shapes. The high damping of the knit mesh results in rapid dissipation of the excitation energy. Since the input energy did not propagate long distances, it was extremely difficult to produce a standing wave (vibration mode) in the surface mesh. To insure that membrane theory was adequate to model the mesh, a 1.22 m square knit mesh model was constructed for vibration testing.

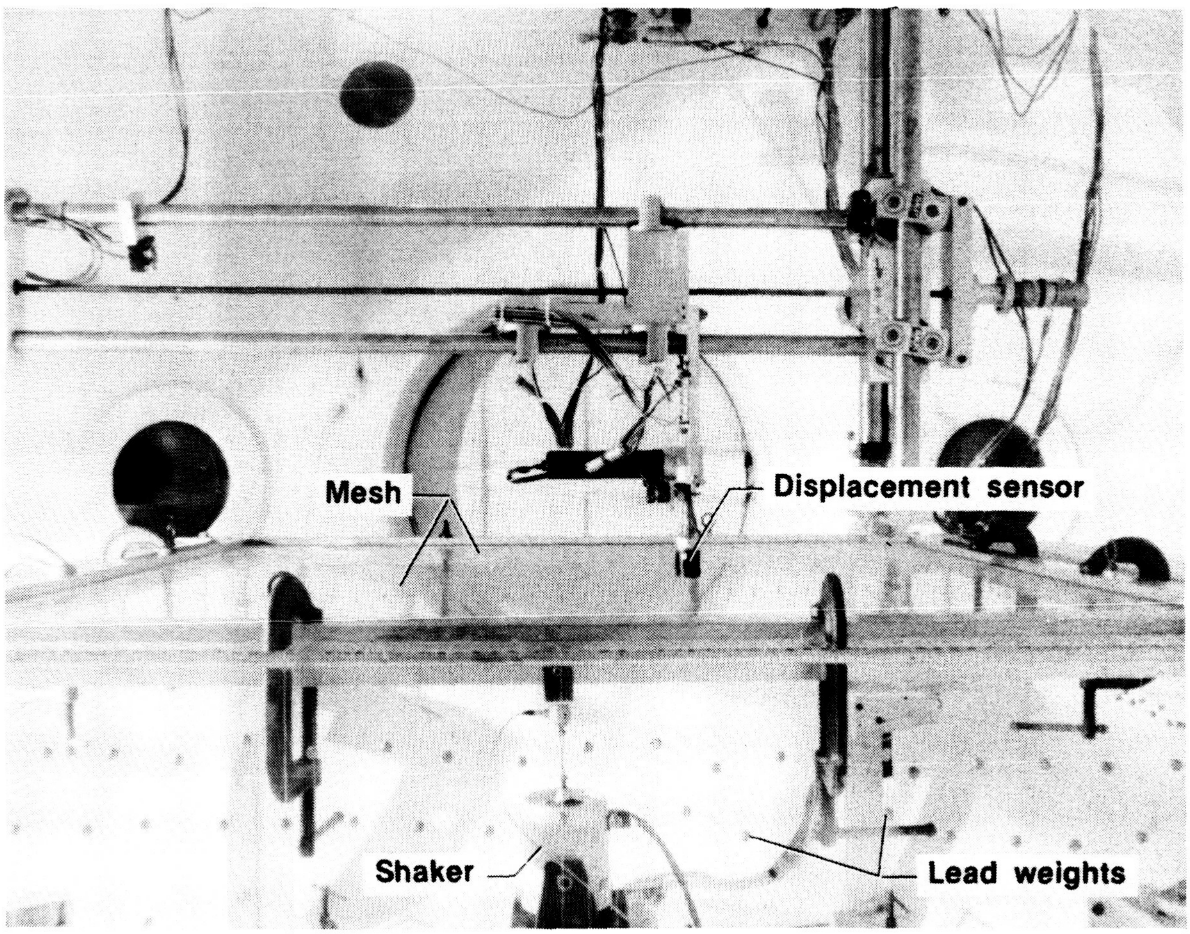
70 modes from 4.1 to 6.2 HZ



Typical mesh modes

1.22 METER SQUARE MESH VIBRATION MODEL

The figure below shows a 1.22 m square sample of the gold plated molybdenum mesh used for the reflector surface on the 15 m antenna model. The model was mounted in an 2.5 m vacuum sphere to permit testing at near vacuum conditions. An electrodynamic shaker was attached to the mesh for excitation and a proximity sensor mounted on a survey system was used to measure the response displacements. The mesh was pretensioned uniformly by lead weights which hung over circular rods. Tests were performed at two different tension levels to study tension effects; 3.01 N/m (0.0172 lbs/in.) and 6.02 N/m (0.0344 lbs/in.). The mesh density was measured to be 0.06284 g/m^2 (8.938 X 10^{-8} lbm/in^2). A differential equation for solution of a two dimensional membrane was used to predict the vibration frequencies.



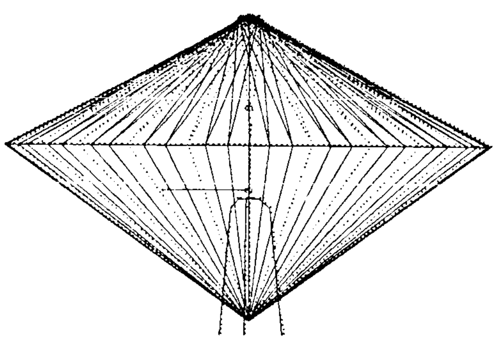
1.22 METER SQUARE MESH DYNAMICS

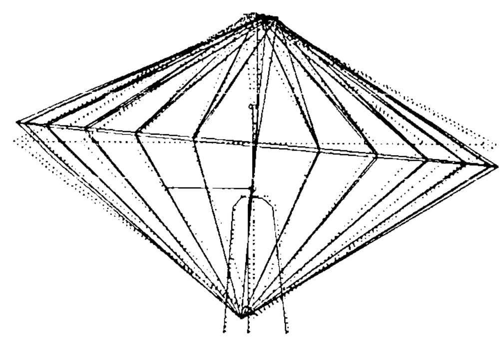
The table below lists frequencies and damping for selected vibration modes of the 1.22 m surface mesh model. The analytical frequencies are high perhaps due to the manner in which the lead weights were used to pretension the mesh. Since the lead weights hung over a circular rod, friction could reduce the amount of tension that is applied to the mesh. Nevertheless, the frequencies increase by the square root of the tension increase which indicates that the mesh can be modeled with membrane theory for prediction of out-of-plane vibration modes. The damping of the mesh is quite high, for example, 6.9 percent of critical damping in the first mode. Ambient air significantly increases the damping of the mesh. The frequencies of vibration are slightly lower at near vacuum pressure than at ambient air pressure. This is unusual since ambient air generally has an apparent mass effect and thus lowers vibration frequencies. One additional point to be made is that the damping tends to decrease as the tension level increases.

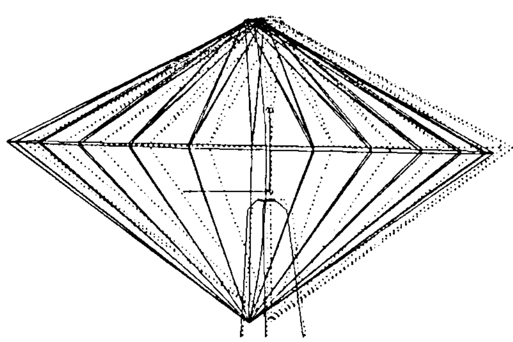
Analysis		Test			
$N_X = N_Y = 3.01$ N/M (0.0172 lbs/in)		Vacuum		Ambient air	
Mode	F (HZ)	F (HZ)	C/CR (%)	F (HZ)	C/CR (%)
1	6.48	5.80	6.9	5.98	9.2
2 (2)	10.24	9.21	4.0	9.35	6.9
3	12.95	13.04	3.3	13.22	4.2
5 (2)	16.51	14.96	3.1	15.17	4.2
$N_X = N_Y = 6.02$ N/M (0.0344 lbs/in)					
1	9.16	8.26	5.0	8.43	7.1
2 (2)	14.48	12.89	4.3	12.96	4.7
3	18.31	16.64	3.1	16.79	3.7
5 (2)	23.35	20.93	2.3	21.13	3.3

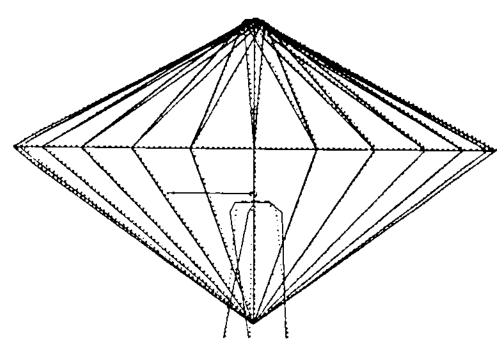
SIMPLIFIED ANALYTICAL MODELS

The dominant modes contributing to antenna vibrations are the global modes described previously. Since these modes are not significantly affected by the surface mesh, a reduced analytical model was developed. The reduced model used a very crude representation of the surface mesh which resulted in only 996 degrees of freedom in the analysis. This model predicted the first four global modes quite accurately as shown in the table below. The mode shapes, also shown below, are similar to those shown previously. The column has been reduced using continuum beam theory (Ref. 6) which makes mode shape plots more difficult to visualize. The good accuracy obtained with the reduced model indicates that relatively crude representations of the surface are adequate to predict the global modes of the antenna.


Mode 1


Mode 2

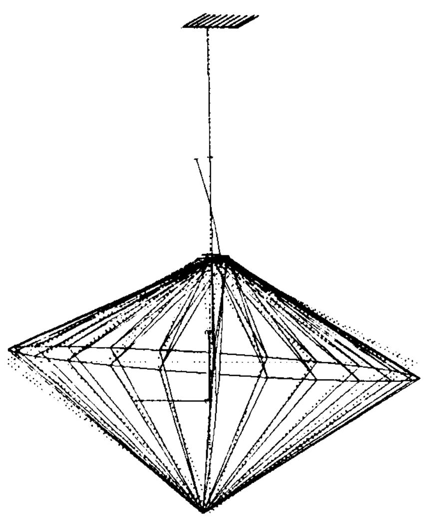

Mode 3


Mode 4

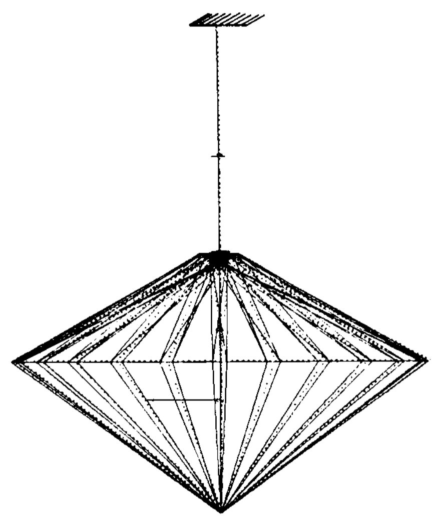
Mode	Full model	Reduced model
	Frequency (HZ)	
1	0.077	0.077
2 (2)	0.697	0.697
3 (2)	1.73	1.84
4	3.20	3.25

ANTENNA MODES WITH CABLE SUSPENSION AND FEED MAST

The reduced model was used to predict the global modes of the antenna when supported by a pendulum cable. A simulated feed weight and feed mast were also attached. The analytical model predicted the cable suspended frequencies shown below. A comparison of the test and analysis results show good agreement. Except for the first two pendulum modes, the cable suspended modes closely agree with the frequencies predicted for free-free boundary conditions. The fundamental flexible antenna frequency increases to 1.47 Hz for cable suspended or free-free vibrations. The test data which was acquired in ambient air shows the damping to be near 1 percent of critical for most modes. Antenna support systems can greatly influence the frequency and mode shapes of the global antenna modes. The hoop and mesh modes are not strongly affected by column support systems thus the hoop and mesh results shown previously are still valid when the antenna is cable suspended or free-free.



Mode 3



Mode 4

Mode	Analysis (reduced model) F (HZ)	Test F (HZ)	Test C/CR (%)	Mode shape
1 (2)	0.132	0.138	3.1	First pendulum
2 (2)	0.283	0.284	1.2	Second pendulum
3 (2)	1.47	1.47	1.3	Hoop rocking/column and feed mast bending
4	2.20	2.19	1.1	Column/hoop torsion
5 (2)	4.37	4.09	1.9	Second column and feed mast bending
6	5.40	5.42	0.94	Feed mast torsion

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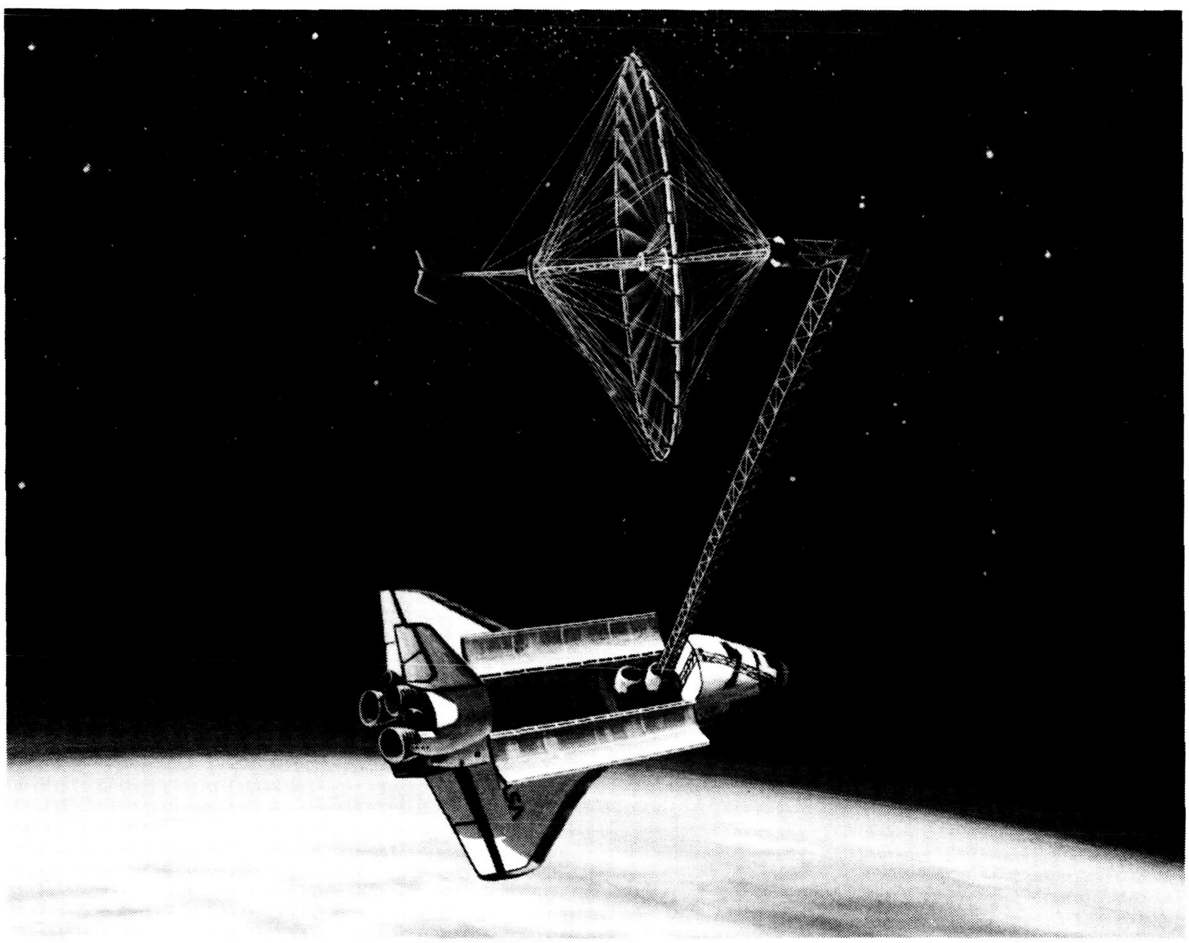
PROPOSED COFS II ANTENNA FLIGHT EXPERIMENT

The 15 meter hoop-column antenna has been tested for deployment, electromagnetic performance and structural behavior. Quasi-static shape control (Ref. 7) has shown the antenna is amenable to active surface shaping. Due to its demonstrated performance and interesting vibration behavior, the antenna is a candidate for flight experiments in the Control of Flexible Structures (COFS) program. The antenna structural behavior has been characterized with two suspension systems, however, the COFS II experiment illustrated below will require further structural analysis and testing. For example, more research is needed to fully understand the interaction between surface shape control and global antenna vibrations.



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SUMMARY

A 15 meter model of the hoop-column antenna concept has been vibration tested for model characterization and analytical model verification. Linear finite element analysis predicted the global vibration frequencies accurately, however, good agreement between analysis and test data was obtained only after the analytical model was refined using static test data. As structures become more flexible, structural properties determined from static data become more accurate and should be used to update analytical models. Global vibration modes are not significantly affected by the surface mesh which permits simplified analytical models to be used for prediction of global behavior. These reduced models are believed sufficient for preliminary design and controls simulations where only global behavior is desired. The mesh modes were highly damped due to the knit mesh used for the reflector surface. These modes were also highly coupled and very difficult to measure in the laboratory. The inability to fully characterize the antenna mesh modes in the laboratory indicates robust methods for active surface vibration suppression will be needed. Fortunately, the surface mesh exhibits high passive damping which should be beneficial to active control systems.

- Static tests are needed to refine analytical models
- Mesh modes are highly damped (coupled)
- Reduced analytical models are adequate to predict global vibration modes
- The effect of cable and mesh modes on active control systems needs further investigation

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